HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS IN A CHANNEL WITH GRIP METAL SURFACES

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ABSTRACT

A common technique for enhancing convective heat transfer is extended surfaces (fins). The size and shape of these fins plays a crucial role in their performance. Much research has investigated ways to leverage novel manufacturing techniques to create different shapes to improve thermal-fluidic performance. NUCAP has developed a proprietary skiving process which creates an array of hook-shaped raised features on metal surfaces (trademarked as GRIP Metal). These features (hooks) offer increased specific surface area and increased fluid mixing and/or promote boundary layer separation which serve to improve convective heat transfer. The hooks have a nominal height, h_f = 1.5 mm and are arranged in a staggered layout. They are applied to the upper and lower planes of a rectangular channel. The present work numerically investigates the thermal and hydraulic performance of this rectangular channel with hooks and compares them to a corresponding flat plate. The effect of changing the clearance, *C*, between the hooks' tip and the opposing plate is examined. All results are obtained by solving a conjugate heat transfer problem for Reynolds number, *Re*, ranges from 500 to 4000 with air as a working fluid. A heat flux ranges from 425 to 3410 W/m² is applied to maintain a consistent temperature rise of the air for all cases. The results show that channels with GRIP Metal hooks have higher thermal performance than that of the flat plate, however, with a penalty in pressure drop. Generally, the heat transfer coefficients and the friction factors for the channel shows a great dependency on the value of the clearance. The channel with a clearance of $C = 0$ has the lowest thermal-hydraulic performance. On the other hand, the channels with clearances of $C = h_f$ and $C = 0.5h_f$ depicts the greatest heat transfer enhancement compared to the flat plate at $Re < 2000$. However, at $Re = 4000$ the channel with $C = 2h_f$ shows the same heat transfer enhancement as the $C = h_f$ but with a lower pressure drop.

INTRODUCTION

Many methods have been developed to enhance convective heat transfer in industrial applications. The concept of finned surface is one of the most frequently used method. They are extended surfaces of different geometries attached to the surface that increase the heat transfer area and improve the turbulent mixing which results in enhancing the heat transfer [1,2]

Examples of fins can be found in heat exchangers used in heating and air-conditioning systems, chemical processing, power generation plants, vehicles, …etc. [1] Also, they have been extensively employed for cooling turbine aerofoils and in electronic devices' cooling to prevent overheating of the system.

Both the heat transfer and the pressure drop associated with the flow over such arrays have been subjected to extensive investigations to capture the flow's characteristics. Then many efforts were exerted to optimize the performance of these arrays by varying their geometries and configurations. It is worth mentioning that these optimization techniques aimed to achieve the maximum possible heat transfer with minimal effect on the pressure drop.

The most common pin fin shape was the cylindrical shape for its superior heat transfer performance and ease of manufacturing [3]. In the beginning, these pin fins had a relatively large height to diameter ratio, h_f/D, i.e., $h_f/D > 4$. However, more recently, with the benefit of microfabrication technologies the geometrical features of the pin fins were shrunk to be just few millimeters, i.e., $h_f/D < 4$. and even smaller to the microscale for cooling applications within compact spaces, such as electronics and turbine cooling [4]. Unfortunately, the heat transfer of such short fins, in which h_f/D is in the order of unity, was lower than their longer correspondents [5,6] and the correlations that represent the heat transfer and the pressure drop for the long fins are not suitable for the short fins [6]. In contrary to the long fins in which the heat transfer from the uncovered region (end-walls) is negligible, the heat transfer from these end-walls in case of the short ones is almost equal to the heat transfer from the fins themselves [7]. It is believed that the heat transfer from these end-walls depends strongly on the geometry of the pin-fins and their distribution [8]. The addition of pin-fins increases the wetted area and produces higher turbulence mixing which subsequently increases the heat transfer, however, this contribution of area addition towards the heat transfer augmentation is low in case of short fins. This pertains to that the covered area by the added fins is comparable to the surface area of the fins themselves. Hence, the turbulence mixing is the dominant factor for the heat transfer augmentation in short fins.

Consequently, it was believed that there could be a pin-fin shape that enhances the turbulence mixing higher than the cylindrical pin-fins. Therefore, many investigations were conducted to examine different pin-fin cross sections. For instance, Metzger et al. [9] compared the heat transfer and the pressure drop characteristics of circular and oblong pin-fins with varying angles of attack experimentally. Their results revealed that the heat transfer of oblong pin-fins is higher than that of the circular pins by a maximum of 20%, however, at the expense of having double pressure drop. Similarly, Chyu et al. [10] studied the heat transfer enhancement induced by an array of cubic/diamond fins compared to the circular one via the naphthalene sublimation technique and heat/mass transfer analogy. They found that the heat transfer for the cubic pin-fins is 40% and 80% higher than that of the diamond and the circular arrays, respectively. While Sahiti et al. [11] numerically investigated the influence of several pin-fin cross-sections (NACA, Drop shape, Lancet, Elliptic, Circular, and square) for both staggered and in-line arrangements on the array's heat transfer performance. Their results showed that either the elliptic or the drop form pin-fins are superior to their peers depending on the geometrical parameters of the array (pin length, transverse and longitudinal spacings and coverage ratio). Another method of enhancing the heat transfer from pin-fin arrays is the introduction of dimples in the array. Chyu et al. [12] experimentally compared the heat transfer performance from a dimpled rectangular channel and a smooth flat plate. The dimples are either circular or teardrop shaped. Both shapes showed higher level of heat transfer than that of the smooth plate by a factor of 2.5. On the other hand, Moon et al. [13] studied the heat transfer and pressure drop of array characteristics of circular dimples at different channel's height. Their results revealed that the heat transfer augmentation due to these dimples with respect to smooth flat plate is approximately constant at value of 2.1. In addition, both pressure drop, and heat transfer are independent of channel's height. Rao et al. [14] introduced the concept of hybrid pin fin-dimple arrays. They conducted an experiment to compare the thermal and hydraulic performance between pin fin-dimple and pin-fin arrays. As well as study the effect of dimple depth on that performance. Their results showed that the presence of dimples improved the heat transfer by up to 19% depending on the dimple depth and *Re* number, while the friction factor is lower by 17.6% for the shallower dimples than that of the pin-fin arrays.

NUCAP Industries has developed a skiving manufacturing process to create high aspect ratio metal features and dimples/cavities on metal surfaces. Due to the nature of the skiving process these features are hook shaped, thus usually referred as hooks in this paper. These hooks offer increased surface area, enhance the turbulent mixing of the fluid and/or promote boundary layer separation that serve to increase connective heat transfer.

The objective of this work is to conduct a comparison between rectangular channel with hooks and a flat plate in terms of the heat transfer and pressure drop characteristics. In addition, the effect of varying the channel's clearance above the hooks (0, $0.5h_f$, h_f, $2h_f$) is investigated.

HOOKS GEOMETRY

A detailed structure of NUCAP's hooks geometry is depicted in **Figure 1** and the nominal values of their different geometrical parameters (height, spanwise pitch, streamwise pitch, etc.) are reported in **Table 1**. These hooks are manufactured by skiving process. In other words, the hooks are partially removed material from the plate. Thus, downstream of each hook there is a dimple/cavity from which the hooks where formed. These dimples are the negative volume of the hooks.

In this paper, the commercial CAD–embedded CFD software provided by Solidworks is used to predict the heat transfer and pressure drop characteristics for rectangular channel with GRIP Metal fins (hooks) and compare them to that of a flat plate at different flow rates. The channel's clearance above fins *C* is also increased from 0 to twice the hooks' height (h_f) to investigate its effect on the performance of the hooks.

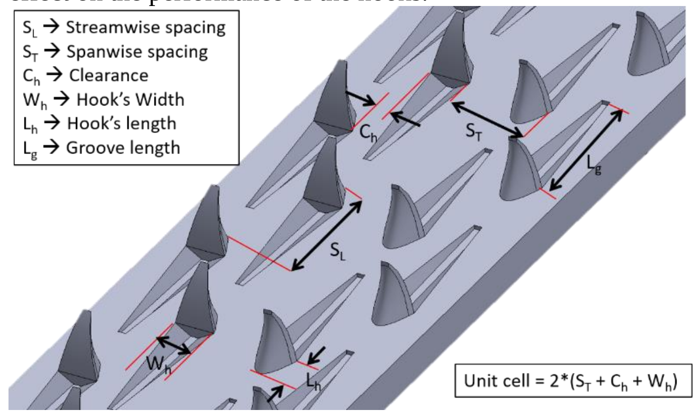

Figure 1 Geometrical parameters of an array of hooks

Table 1 Nominal values of the geometrical parameters of an array of hooks

h_f (mm)	S_L (mm)	S_T (mm)	W_h (mm)	C_f (mm)	L_h (mm)	L_g (mm)
1.52	3.175	2.286	1.016	0.381	0.826	4.140

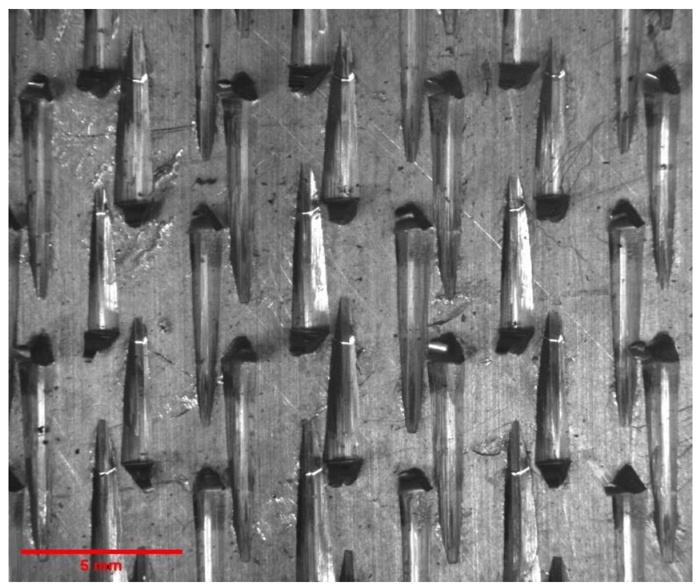

Figure 2 Magnified image of aluminum plate with NUCAP's hooks.

PHYSICAL MODEL

Since the hooks array is repetitive in the spanwise direction as depicted in **Figure 2**. A geometrical model, shown in **Figure 3**, of the hooks is developed using their nominal data. This model represents a unit cell of the array which is periodic in the spanwise direction to save the computational time.

Figure 4 shows the whole computational domain studied for the rectangular channel of $C = 2h_f$. This domain can be divided into three main regions in the flow direction. Firstly, the developing section of length, L_d, equal to the maximum required developing length i.e., the developing length at highest *Re* number. This developing section ensures that the flow over the hooks is hydrodynamically fully developed. Secondly, the hooks section of length, L_f, equals 4.125 inches, in which there are 33 rows of hooks in the streamwise direction arranged in staggered layout with height, h_f, of 1.5 mm. Finally, the mixing region of length, L_m 25% that of the developing length downstream of the test section in which the fluid is settled down. The rectangular channel has a height, H_c, of 1.5 mm, 2.25 mm, 3 mm, or 4.5 mm, which corresponds to clearance above hooks of 0, 0.5h_f, h_f, 2h_f respectively. A representation and dimensions for each rectangular channel are shown in **Figure 5** and **Table 2**, respectively.

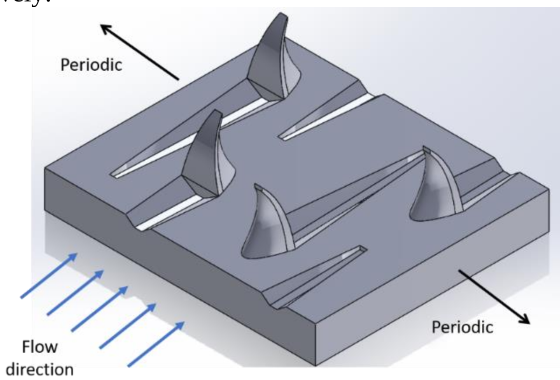

Figure 3 Geometrical model of a unit cell for NUCAP's hooks array

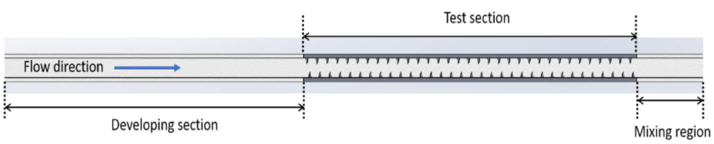

Figure 4 Computational domain for C = 2h_f

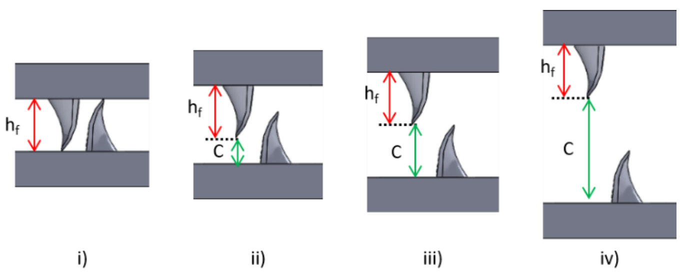

Figure 5 Rectangular channels with different values of C
i) C = 0 - ii) C = 0.5h_f - iii) C = h_f - iv) C = 2h_f

Table 2 Dimensions of the tested rectangular channels

C	0	0.5h_f	h_f	2h_f
H				

BOUNDARY CONDITIONS

While the fluid across fins' array is a typical example of external convective heat transfer, the small height of the rectangular channel prohibits the developing of thick boundary layer which gives the flow the characteristics of the internal flow. Thus, the above-mentioned computational domain is solved as a steady state, 3D internal incompressible flow with heat transfer. Air is implemented as working fluid. It is assumed to be an ideal gas and hence the density is pressure and temperature dependent while other properties (specific heat, thermal conductivity, dynamic viscosity) are only temperature dependent.

A constant temperature, T_{in}, of 20 °*C* is applied at the inlet of the channel. While the inlet velocity, V_{in}, is varied to obtain Reynolds number in the range of 500 and 4000 with turbulence intensity of 5%. The outlet static pressure of the channel is set to be 0 Pa for all cases and back flow of 20 °*C*. Periodic conditions are applied to the side walls of the domain. Constant heat flux, q_s, is applied to bottom and the upper surfaces of the test section. The heat flux is adjusted to maintain a temperature rise of 20 °*C* in air across the domain. No slip boundary conditions are applied to all other walls with adiabatic conditions.

DATA REDUCTION

For a conjugate heat transfer problem, the aluminium plates tend to have a constant surface temperature through axial conduction resembling a constant temperature boundary condition in which the fluid temperature approaches the wall temperature T_w exponentially [15]. Thus, logarithmic temperature difference ΔT_{lm} is adopted and calculated as:

$$\Delta T_{lm} = \frac{(T_{w,in} - T_{in}) - (T_{w,o} - T_o)}{ln(T_{w,in} - T_{in}) - ln(T_{w,o} - T_o)} \tag{1}$$

where $T_{w,in}$ and $T_{w,o}$ are the inlet and outlet surface temperature, respectively. And T_o is the mass weighted average temperature of the fluid at the outlet surface which is calculated by:

$$T_o = \frac{\int \rho V T \, dA_o}{\int \rho V \, dA_o} \tag{2}$$

where *T* is the temperature, *V* represents the velocity in the flow direction and ρ is the density at each cell of the outlet surface.

Then the averaged heat transfer coefficient of the hooks h_{avg} is calculated based on the following equation:

$$h_{avg} = \frac{\dot{q}_s}{\Delta T_{lm}} \tag{3}$$

where q_s denotes the heat flux from the hooks and the endwall to the air.

It is common to present the heat transfer results in the dimensionless form of Nusselt number *Nuh*:

$$Nu_h = \frac{h_{avg} D_h}{k_{in}} \tag{4}$$

where k_{in} is the thermal conductivity of air at inlet conditions which is equal to 0.02514 W.m^{-1}. K^{-1}, while D_h is the hydraulic diameter of the channel which is equal to:

$$D_h = 2H_c \tag{5}$$

For the pressure drop characteristics, the friction factor f_h is calculated from the following equation:

$$f_h = \frac{2\ \Delta P\ D_h}{L_f\ V_{in}^2\ \rho_{in}} \quad (6)$$

where ρ_{in} is the density of air at inlet conditions which is equal to 1.204 Kg.m^{-3} and ΔP is the difference between the mass weighted average pressure at hooks section's inlet, P_{in}, and outlet, P_o.

Finally, Reynolds number, *Re*, is calculated by:

$$Re = \frac{\rho_{in}\ V_{in}\ D_h}{\mu_{in}} \quad (7)$$

where μ_{in} is the dynamic viscosity air at inlet conditions which is equal to 1.825*10^{-5} Pa.s.

For a comprehensive assessment of both heat transfer and pressure drop characteristics of the hooks compared to those of flat plate, the overall thermal performance η - proposed by Gee and Webb [16]- is evaluated:

$$\eta = \frac{(Nu_h/Nu_o)}{(f_h/f_o)^{1/3}} \quad (8)$$

where Nu_o and f_o are the Nusselt number and the friction factor for a flat plate predicted at the same boundary conditions as of the hooks.

TURBULENCE MODEL AND CONVERGANCE CRITIRIA

Many investigated the flow regime over bank of tubes, which were aiming to identify the critical *Re* after which the flow is no longer laminar and begins its transition to turbulent. However, many observations were stated depending on the streamwise and spanwise distances between the fins and the arrangement in which these tubes are arranged. For instance, Zakausakas [17] observed that the flow over the bank is considered laminar for Re < 1000. While Bergelin et al [18] stated that the flow is laminar if Re < 200 and considered to be in transition regime afterwards until Re = 5000.

In this study *Re* ranges from 500 to 4000, in other words both laminar and transition regimes exist. However, for Re = 500 the solution never converges when the laminar model is applied, which means that the laminar model no longer fits the physics of the flow and the flow regime is transition. Thus, a modified *k-ε* turbulence model with damping functions proposed by Lam and Bremhorst [19] is utilized to achieve the required convergence criteria. The *k-ε* turbulence model turbulence model was shown to be suitable to predict the physics of the flow around pin fins with a reasonable accuracy up to 7% error at Re = 50000 [11,20–22].

The convergence criteria are set to be 10^{-5} to all the residuals of the mass, energy and momentum balances between the inlet and the outlet, and the solution is set to be converged only if all residuals reach this value. Another method for checking the solution's convergence is monitoring the behaviour of important parameters such as the bulk outlet temperature, inlet pressure and surface temperature of the heated section. Then making sure that these parameters reach a constant value before the convergence criteria were satisfied.

NUMERICAL MESH AND GRID INDPENDENCY

SolidWorks Meshing is employed to mesh the current computational domain into hexahedral cells; **Figure 6**.

For grid independence assessment, Nu_h and f_h are predicted with different element sizes at the highest *Re*, which is 4000. As shown in **Figure 7**, the difference in Nu_h between the coarsest mesh (0.75 million cells) and the finest one (3.8 million cells) is 12 %, while this difference decreases to 9 % for f_h as in Figure 8. Thus, a conservative value of 3 million cells is chosen as it has almost the same values for Nu_h and f_h as the finest mesh, however, with a lower number of cells which results in lowering the computational time.

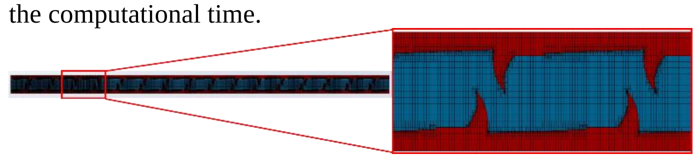

Figure 6 Meshing details of the fluid (Blue) and solid domains (Red)

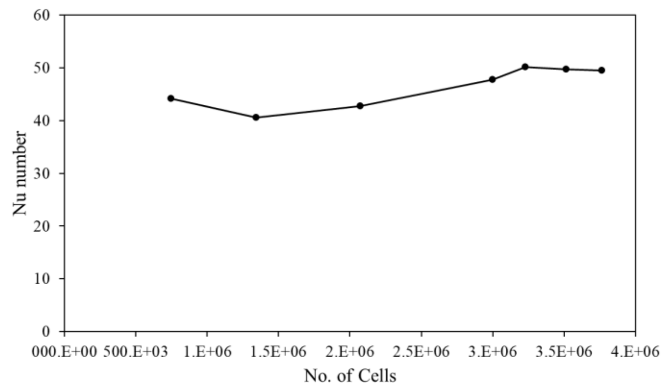

Figure 7 Nu number for different grid sizes at Re = 4000

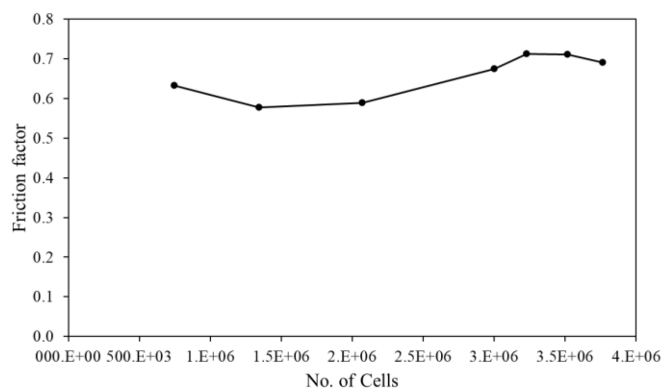

Figure 8 Friction factor for different grid sizes at Re = 4000

NUMERICAL VALIDATION

The validation of the numerical model was performed by comparison of the predicted results with experimental data. For the present study, it was not possible to find experimental data which fit the present *Re* range. Besides, these hooks were not investigated before. Thus, an experimental facility, which is a pre-designed open circuit wind tunnel operating in forced mode was built to validate the current numerical model. The wind tunnel is sub-divided into six sections (fan section. two contraction nozzles, orifice section, developing length section, test section). Each section will be 3D-printed using PLA filaments, then these sections will be assembled. The last two sections (developing length and test sections) are rectangular channels with a width of 2 inches and a height of 3 mm i.e., $C = h_f$. Due to design limitations the lowest *Re* obtained experimentally is 2000, so the validation is only done for $Re > 2000$. Two validations were carried out, the first was for a flat plate whereas the second was for the channel with hooks.

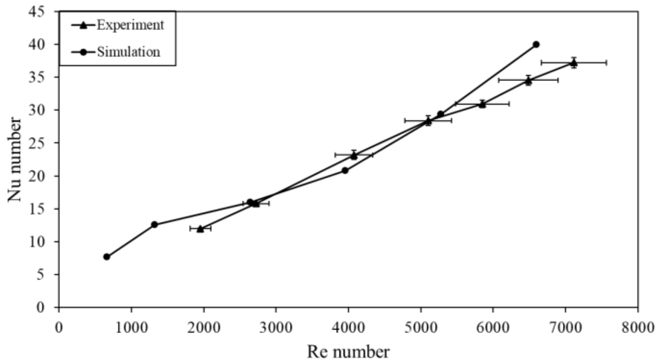

Figure 9 Comparison between Nu number predicted numerically for the flat plate and their corresponding values obtained from the experiments.

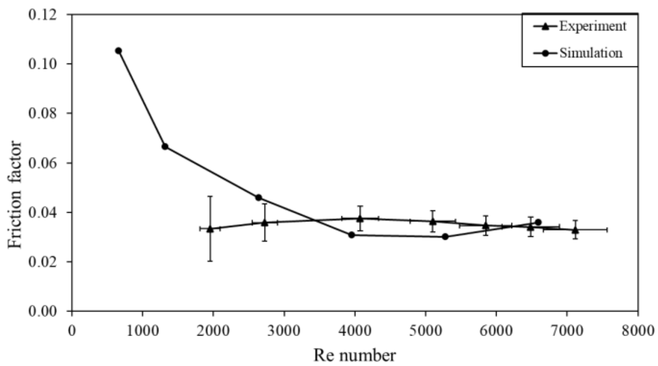

Figure 10 Comparison between friction factor predicted numerically for the flat plate and their corresponding values obtained from the experiments.

Figure 9 & **Figure 10** demonstrate the comparison between the Nusselt number, Nu_o, and the friction factor, f_o, predicted numerically and their corresponding experimental results for a flat plate. The simulation results agree with experiments and the average discrepancy between two series of results is less than 10%. While **Figure 11** & **Figure 12** display the same comparison but for the channel with hooks at $C = h_f$. The maximum deviation of the numerical and experimental data for Nu_h was 19%, whereas the maximum deviation for f_h was 46% observed at high *Re* and 5% at low *Re*.

It is claimed that the agreement of the results is satisfactory to perform comparisons of various channels considering the deviation of the manufactured array of hooks from the numerical model due to the nature of the skiving process. In addition, the numerical model considers the flow as fully turbulent, although in practice the *Re* lies in the transition between laminar and turbulent.

RESULTS AND DISCUSSION

For the comparison of the heat transfer and pressure drop characteristics of between the channel with hooks and a flat plate, the ratio of Nusselt number for different channels with hooks, Nu_h, to Nusselt number for flat plate at the same boundary conditions, Nu_o, which represents the augmentation factor is plotted in **Figure 13**. This ratio is always greater than 1 which indicates that the presence of these hooks enhanced the heat transfer capabilities of the channel. This enhancement is contributed to the hydrodynamic enhancement of the flow due to the presence of the hooks and their offered increased surface area. An apparent pattern could be found that, this ratio increases with increasing *Re* until it reaches a maximum value of 3.25 and 3.46 at $Re = 2000$ for both $C = h_f$ and $C = 0.5h_f$ then it decreases to become 2.56 and 2.32, respectively. While for the other two clearances this ratio always increases with *Re* such that it reaches a value of 3.14 for $C = 2h_f$, outperforming all other cases, and 2.27 for $C = 0$.

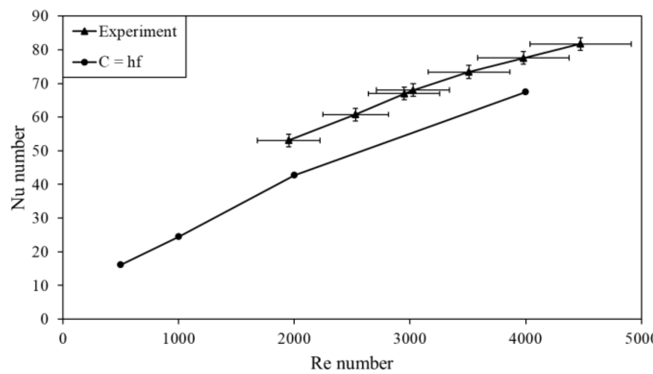

Figure 11 Comparison between Nu number predicted numerically for the channel with hooks and their corresponding values obtained from the experiments.

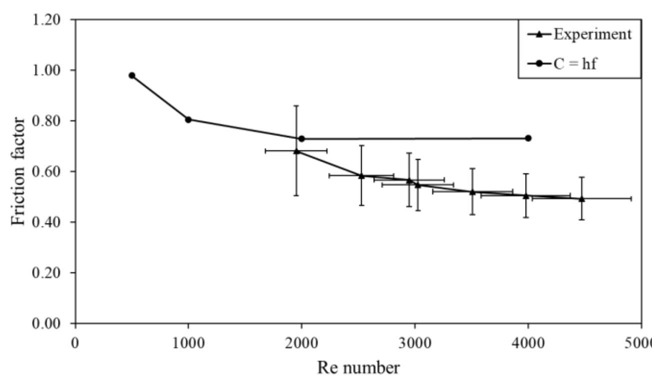

Figure 12 Comparison between friction factor predicted numerically for the channel with hooks and their corresponding values obtained from the experiments.

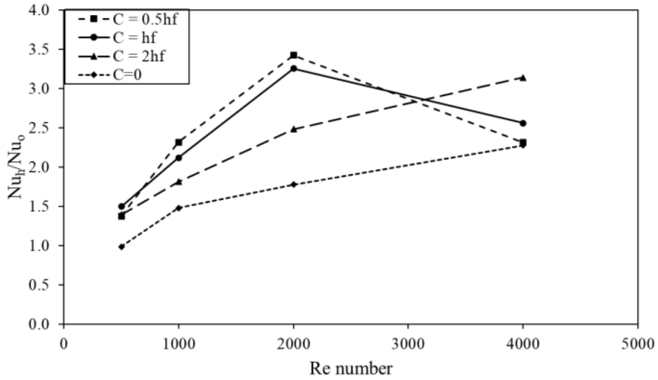

Figure 13 Nu_h/Nu_o vs Re at different values of C.

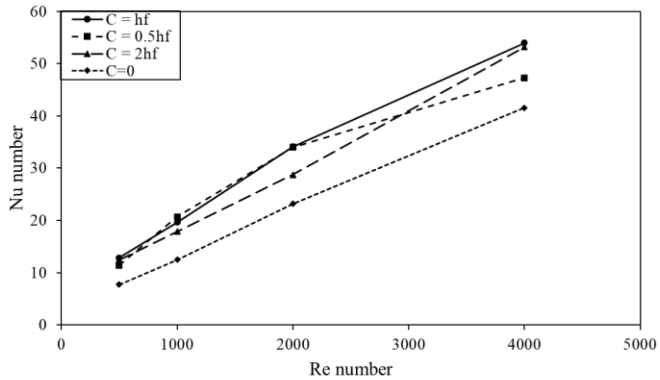

Figure 14 Nu number for channel with hooks, Nu_h, vs Re at different values of C.

For further investigation of the effect of C on the thermal performance of channels with hooks, **Figure 14** shows the Nu_h for channels with hooks at different values of C vs Re. It is noted that for all values of C, Nu_h increases with increasing Re. Also, the channel of C = 0 has the lowest performance which is believed to be related to the strong restriction imposed by the interdigitated hooks to the fluid flow. With the exception of Re = 4000, the $C = h_f$ channel is superior to other channels. While at Re = 4000, the $C = 2h_f$ channel has the same Nu_h value as $C = h_f$ which indicates that at higher values of Re this C value might outperform the other cases.

On the other hand, the ratio of friction factor for different channels with hooks, f_h, to that for flat plate at the same boundary conditions, f_o, is plotted in **Figure 15**. This factor represents the penalty gained in terms of pressure drop due to the presence of hooks. This ratio is between 2.5 and 12.7 which implies that although the presence of these hooks enhances the heat transfer, there is a tremendous increase in the pressure drop. This factor, f_h/f_o, increases as Re increases such that the highest penalty is for $C = h_f$ channel at Re = 4000.

Figure 16 displays f_h for channels with hooks for different values of C at different Re. It is noted that f_h shows a declining trend with increasing Re, and this trend has become almost flat when Re > 2000. Another feature could be concluded that the existing 4 channels can be classified as two groups in which the members of each group has almost the same f_h. The channels with $C = h_f$ and $C = 0.5h_f$ represent the first group, that has the higher values of f_h at all Re, whereas the two remaining channels represent the second one. The latter has values of f_h that is 20% - 29% lower than the former depending on the value of Re. Although higher value of C means wider channel and thus, lower resistance to the flow, both $C = h_f$ and $C = 0.5h_f$ channels have higher friction factor than that of C = 0. On the other hand, $C = 2h_f$ channel shows a relatively lower friction factor values than the $C = h_f$ and $C = 0.5h_f$. This phenomenon can be contributed to the fact that as increasing C allows more air to bypass the fins and flow through the low resistance gap between the two plates decreasing the skin friction between the hooks and the air.

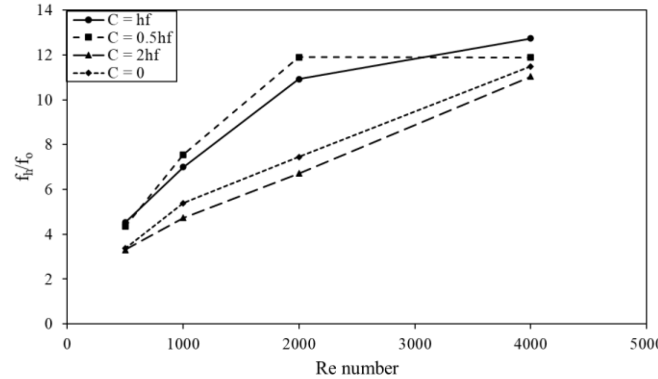

Figure 15 f_h/f_o vs Re at different values of C.

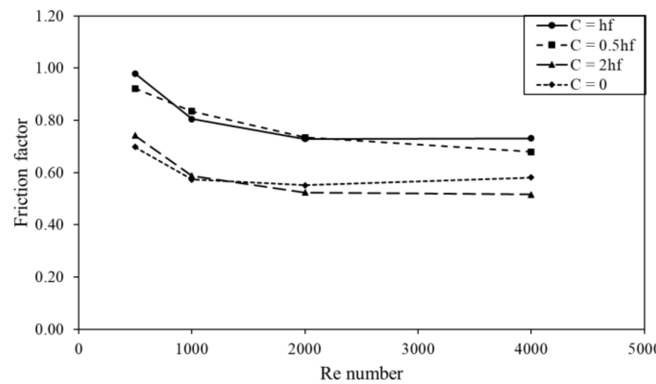

Figure 16 Friction factor for channel with hooks, f_h, vs Re at different values of C.

Finally, the overall thermal performance factor, η, is evaluated and shown in **Figure 17**. It is found that at Re = 500 the value of this performance factor for all values of C is less than 1, which illustrates that the enhancement in the heat transfer does not outweigh the pressure drop penalty. In addition, η for C = 0 and $C = 2h_f$ channels gradually increases as Re increases, while for the other two channels i.e., $C = h_f$ & $C = 0.5h_f$, η increases to a maximum value of 1.5 at Re = 2000 then decreases dramatically to a value of 1 at Re = 4000. The value of η for the channel of C = 0 is the lowest among the present channels and always less than 1, whereas, in general, the channels of $C = h_f$ & $C = 0.5h_f$ outperform the other two at all values of Re except for Re = 4000, at which the channel of $C = 2h_f$ takes the lead with a value of η equals 1.4.

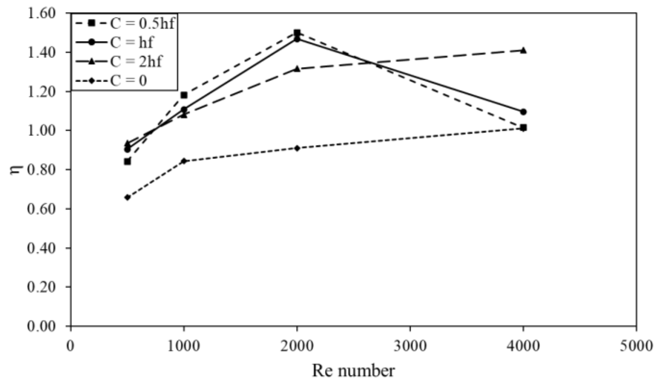

Figure 17 Thermal performance factor for channel with hooks, η, vs *Re* at different values of *C*.

CONCLUSION

In this study, the thermal and hydrodynamic performances of an array of hook-shaped raised features on metal surfaces developed by NUCAP (trademarked as GRIP Metal) is assessed and compared numerically to that of a flat plate. These features, hooks, offer increased surface area and increase fluid mixing and/or promote boundary layer separation which serve to increase connective heat transfer, however, with an inevitable penalty of increased pressure drop. Besides, the effect of changing the channel's clearance above the hooks is investigated. Based on the obtained results, these novel hooks showed a potential to emerge as heat transfer enhancement technique among the different available ones and the following conclusions can be drawn:

- CAD-embedded CFD software provided by SolidWorks which utilizes a modified k-ε turbulence model with damping functions underpredicts the Nu_h by 19%, whereas it overpredicts the f_h by 46% and 5% at high and low *Re*, respectively.
- In general, the rectangular channel occupied with these arrays of hooks outperformed the conventional flat plate by offering a higher *Nu* for the whole studied range of *Re*, nevertheless, with a penalty of adding more pressure drop.
- The ratio between the channel's clearance above the hooks to their height has an intensive influence on the performance of the studied array both thermally and hydrodynamically. However, one can conclude that Nu_h increases with increasing *Re* for all values of *C*, whereas f_h decreases gradually reaching a constant value at *Re* = 2000.
- The maximum enhancement in the heat transfer occurs at *Re* = 2000 for channel of $C = 0.5h_f$, at which the Nu_h is 3.46 times that of a flat plate at the same *Re*.
- However, this enhancement is accompanied with increase in the pressure drop in such a way that at *Re*=4000, the f_h of the array of hooks is 12 times that of a flat plate for $C = h_f$ channel.
- The overall thermal performance, η of the rectangular channels with hooks is greatly affected by the ratio between the channel's clearance above the hooks to their height and *Re*. For instance, at $C = h_f$ & $C = 0.5h_f$, η has a maximum value of 1.5 at *Re* = 2000, while the table is turned at *Re* = 4000 such that η become maximum at 1.4 for $C = 2h_f$.

The future direction of this research is to conduct further experimental studies to evaluate the accuracy of the numerical results obtained, investigate more configurations of such hooks with different geometrical parameters, and allow for further exploration of novel enhancement geometries.

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